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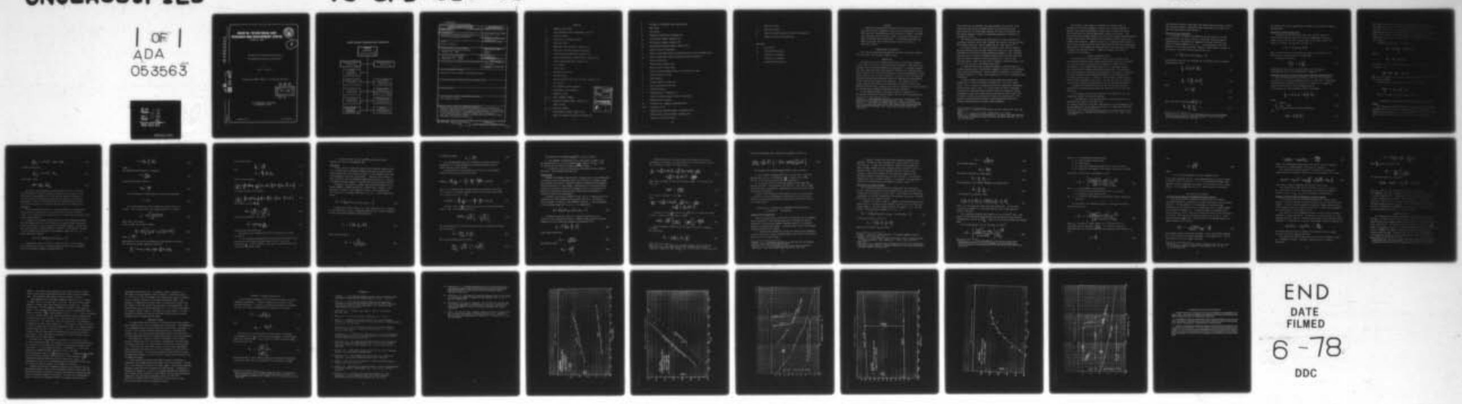

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DAVID W. TAYLOR NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER

Bethesda, Md. 20084

SIMILARITY-LAW CHARACTERIZATION METHODS

FOR ARBITRARY HYDRODYNAMIC ROUGHNESSES

by

Paul S. Granville

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SHIP PERFORMANCE DEPARTMENT
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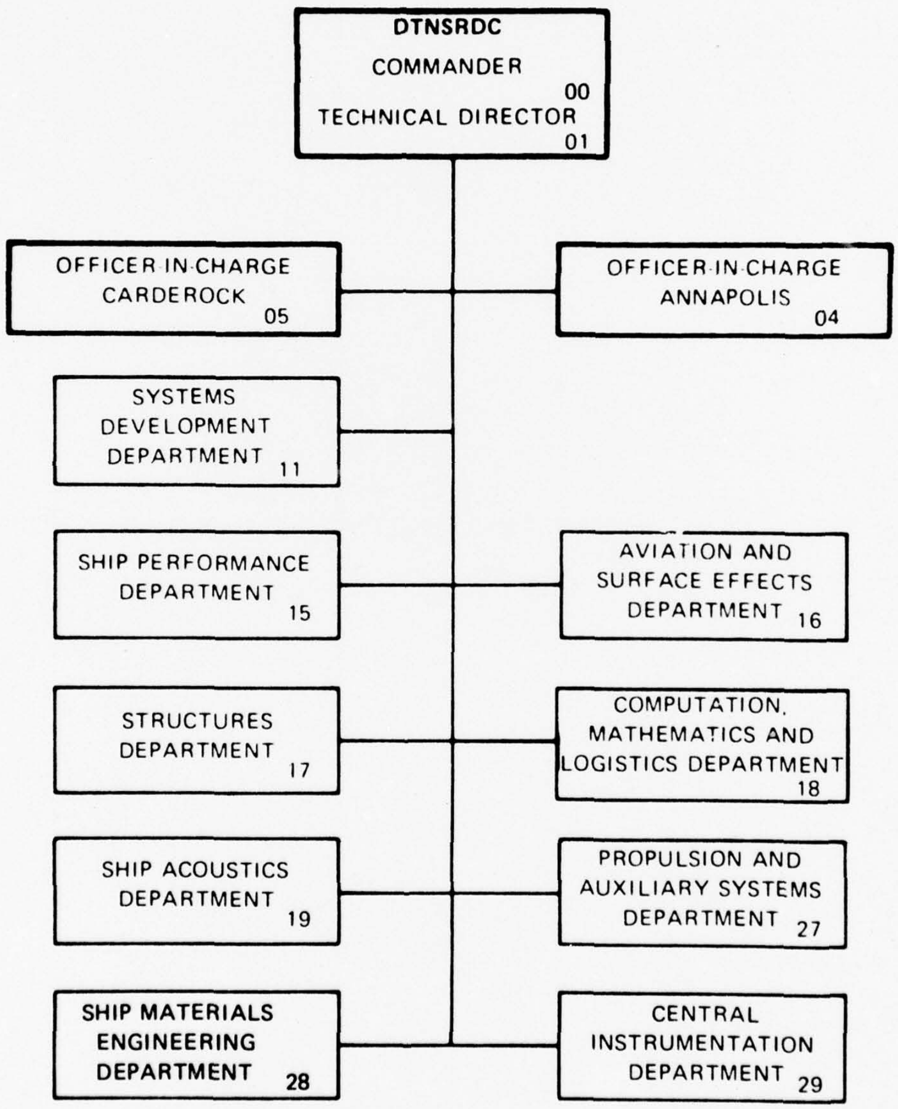

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Methods are described for obtaining similarity-law characterization for arbitrary roughnesses indirectly from pipe-flow head loss, flat-plate drag, and rotating-disk torque. The roughness characterizations are used to make viscous drag predictions by means of boundary-layer calculations. A simplified drag prediction method using form factors and equivalent rough flat plates is described in detail.

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NOTATION

A	Boundary-layer factor
$\tilde{A}$	Cross-sectional area of underwater ship hull
B_1	Inner law factor
$B_{1,0}$	Value of B_1 for smooth surfaces
B_2	Outer law factor
B_r	Roughness inner law factor, Equation (13)
$B_{r,0}$	Value of B_r for fully developed roughness
C_F	Flat-plate drag coefficient, Equation (35)
C_m	Moment coefficient for rotating disk, Equation (44)
C_v	Viscous drag coefficient, Equation (54)
D	Pipe diameter
D_v	Viscous drag
D_1	Boundary-layer factor
$\tilde{D}$	Flat-plate drag
f	Fanning friction factor for pipe flow, Equation (21)
$\tilde{f}$	Form factor
H	Two-dimensional shape parameter
h	Axisymmetric shape parameter
k	Roughness height
k_1, k_2	Other roughness lengths
k^*	Roughness Reynolds number, Equation (5)
L	Length of body
L_1, L_2	Other body lengths
L^*	Nondimensional length, Equation (67)
M	Moment or torque of oneside of rotating disk

$\tilde{P}$	Perimeter of underwater hull cross section
q	Wake factor
R	Disk radius
$\tilde{R}$	Roughness configuration, Equation (6)
R_D	Pipe Reynolds number, Equation (22)
R_L	Body Reynolds number, Equation (36)
R_R	Rotating-disk Reynolds number, Equation (45)
r	Radius of body of revolution
r_A	Radius of equivalent body of revolution based on cross-sectional area
r_p	Radius of equivalent body of revolution based on perimeter
S	Wetted surface area
$\tilde{S}$	Shape of body, Equation (56)
u	Mean velocity in shear layer
U	Velocity outside boundary layer or at centerline of pipe
U_∞	Forward velocity of body
u_τ	Shear velocity
V	Average velocity of pipe flow
x	Axial distance from body nose
y	Distance from wall
y^*	Nondimensional y, Equation (4)
α	Angle of meridian contour of body of revolution
δ	Boundary-layer thickness
ΔB	Similarity-law roughness characterization
$(\Delta B)'$	Derivative of ΔB
η	Boundary-layer Reynolds number, see Equation (25)
λ	Darcy-Weisbach friction factor, Equation (23)
ν	Kinematic viscosity of fluid

ρ	Density of fluid
τ_w	Wall shear stress
ϕ	Angular velocity factor for enclosed rotating disk
ω	Angular velocity of rotating disk

Subscripts

e	end of body
en	enclosed rotating disk
r	condition of roughness
s	condition of smoothness
∞	unenclosed rotating disk

ABSTRACT

Methods are described for obtaining similarity-law characterizations for arbitrary roughnesses indirectly from measurements of pipe-flow head loss, flat-plate drag, and rotating-disk torque. The roughness characterizations are used to make viscous-drag predictions by means of boundary-layer calculations. A simplified drag-prediction method using form factors and equivalent rough flat plates is described in detail.

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INTRODUCTION

The prediction of the increase in drag due to a particular roughness on ships does not seem to be a straightforward procedure, especially for a roughness with an arbitrary geometrical configuration. One paper advocates just full-scale ship tests[1] while another[2] considers only deviations from smooth conditions for a towed friction plane with various roughnesses without further analysis for full-scale conditions. The role of the similarity-law roughness characterization does not seem to be understood in extrapolating results obtained under model-scale conditions. The prediction of the viscous drag of rough bodies by any boundary-layer analysis, even at the equation-of-motion level, requires as an input a similarity-law characterization for the particular roughness configuration being considered.

At high Reynolds numbers, the primary effect of roughness is in increasing turbulent skin friction which is the case to be considered here.

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[1] Johnsen, S., "Drag Reducing Coatings for Ship Hulls" presented to 8th Skandinaviska Lackteknikers Forbund Congress, 29 Sep - 1 Oct 1976.

[2] West, E.E., "The Effect of Surface Preparation and Repainting Procedures on the Frictional Resistance of Old Ship Bottom Plates as Predicted from NSRDC Friction Plane Model 4125," Naval Ship R&D Center Report 4084 (May 1973).

Since the physics of turbulent flow and turbulent skin friction is not sufficiently developed to provide theoretical predictions for even smooth surfaces, let alone rough surfaces, the velocity similarity laws have proved immensely valuable in supplying engineering solutions on a semi-empirical basis.

The velocity similarity laws have proved to be very successful in correlating turbulent shear flows, internal and external, such as pipe flow and boundary-layer flow, and for smooth and for rough surfaces.[3,4]

For rough surfaces, the velocity similarity laws lead to roughness characterizations which have been verified in principle by the classical experiments by Nikuradse[3] on dense sand-grain roughness. Here, different size sand grains in pipes of different diameters gave friction losses and velocity profiles which are closely correlated by a similarity-law characterization. Hama[5] has shown that a roughness characterization is the same for both pipe flow and boundary-layer flow.

An inherent problem with irregular roughness is the difficulty of actually manufacturing a reduced-scale roughness if desired to suit reduced-scale laboratory experiments. The most feasible procedure is to treat an irregular roughness as a full-scale roughness per se even in laboratory-scale experiments.

A direct roughness similarity-law characterization requires the measurement of a velocity profile correlated with a local wall shear stress. This is usually an arduous and difficult undertaking, especially when the shear layer is thin. It is much easier to obtain a roughness similarity-law characterization indirectly from more convenient overall measurements such as the average velocity and pressure drop in pipe flow, the drag of a towed flat plate and the torque of a rotating disk. The indirect characterizations, however, require appropriate analytical relations derived from the similarity laws themselves.

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[3] Schlichting, H., "Boundary-Layer Theory," 6th Ed., McGraw-Hill, New York, 1968.

[4] White, F.M., "Viscous Fluid Flow," McGraw-Hill, N.Y. (1974).

[5] Hama, F.R., "Boundary-Layer Characteristics for Smooth and Rough Surfaces," Transactions of Society of Naval Architects and Marine Engineers, Vol. 62, pp. 333-358 (1954).

The purpose of this paper is to present and assemble explicit procedures for obtaining roughness similarity-law characterizations indirectly from overall measurements of pipe flows, of towed flat plates, and of rotating disks based on appropriate analytical relations. The presentation also includes methods of utilizing the roughness characterizations to make predictions of the viscous drag of underwater bodies from various boundary-layer analyses. Emphasis is given to a simplified method based on form factors and equivalent flat-plate drag charts for particular roughnesses. Simple methods are presented for preparing these flat-plate drag charts from similarity-law characterizations. Numerical examples are given to illustrate the procedures.

VELOCITY SIMILARITY LAWS

The present state of development of the velocity similarity laws for turbulent shear flows past irregular rough surfaces are first reviewed.[6] The classical velocity similarity laws developed by Prandtl and by von Kármán for flow through smooth pipes led to a logarithmic formula which successfully correlated existing frictional data. The similarity laws were extended to rough pipes by Nikuradse who successfully correlated the frictional data of dense sand-grain roughnesses of different sizes in various diameter pipes. A variety of other roughness configurations have also been investigated since then. In time, the velocity similarity laws have been extended to other shear flows like boundary-layer flows.

In general for turbulent shear flows the two laws which provide similarity to the mean velocity profile by relating it to the wall shear stress are:

1. The inner law or the law of the wall which applies to the flow immediately adjacent to the solid boundary.

2. The outer law or the velocity defect law which applies to the remaining outer region of the shear flow.

As originally analyzed by Millikan, consideration of an overlapping of the two similarity laws in the shear flow results in a logarithmic functional

[6] Granville, P.S., "The Frictional Resistance and Turbulent Boundary Layer of Rough Surfaces," Journal of Ship Research, Vol. 2, No. 3, pp. 52-74 (Dec 1958).

form for both similarity laws within the common region of overlap. Outside the overlapping region, a functional form is provided by the law of the wake and a wake modification function.[7]

Inner Law or Law of the Wall

A similarity law may be developed for the mean velocity u of the turbulent flow in the region close to a rough wall with an irregular configuration by considering the roughness of the wall defined by any number of length parameters k, k_1, k_2, ...as an addition to the usual parameters of wall shear stress τ_w, density ρ and kinematic viscosity of the fluid ν, and normal distance from the wall y or

$$u = f\left[y, \tau_w, \rho, \nu, k, k_1, k_2, \cdots\right] \tag{1}$$

By dimensional analysis, the variables may be grouped into the following nondimensional ratios:

$$\frac{u}{u_\tau} = f\left[y^+, k^*, \tilde{R}\right] \tag{2}$$

or

$$\frac{u}{u_\tau} = f\left[\frac{y}{k}, k^*, \tilde{R}\right] \tag{3}$$

where

$$y^* = \frac{u_\tau y}{\nu} \tag{4}$$

(5)

u_τ is the shear velocity, $u_\tau = \sqrt{\tau_w/\rho}$ and

$$\tilde{R} = \frac{k}{k_1}, \frac{k_1}{k_2}, \cdots \tag{6}$$

[7] Granville, P.S., "Similarity-Law Entrainment Method for Two-Dimensional Turbulent Boundary Layers in Pressure Gradients," David Taylor Naval Ship R&D Center Report 4657 (Dec 1975).

are dimensionless ratios representing the shape of an irregular roughness configuration.

Outer Law or Velocity Defect Law

In general for turbulent shear flows, the velocity defect U-u has been experimentally found to be directly independent of viscosity, except close to the wall and a function only of wall shear stress τ_w, density of fluid ρ, and distance inward δ-y or

$$U - u = f\left[\tau_w, \rho, y, \delta\right] \tag{7}$$

Here U is the velocity at the center of a pipe or the velocity at the outer edge of a boundary layer of thickness δ.

By dimensional analysis

$$\frac{U-u}{u_\tau} = f\left[\frac{y}{\delta}\right] \tag{8}$$

represents the outer law or velocity defect law which has been experimentally found to be unaffected by a rough wall.[3,5]

Logarithmic Velocity Law and Similarity-Law Roughness Characterization

Both the inner and the outer similarity laws may be considered to be effective in a common overlapping region of a shear flow. Equating the y-derivatives of the inner and the outer laws results in a logarithmic relation for both laws in the overlapping region.[6]
For the inner law, the logarithmic velocity law becomes

$$\frac{u}{u_\tau} = A \ln y^* + B_1\left[k^*, \tilde{R}\right] \tag{9}$$

where

A = constant

$$B_1 = B_{1,0} + \Delta B \tag{10}$$

$B_{1,0}$ = constant value of B_1 for smooth walls

and

$$\Delta B = f\left[k^*, \tilde{R}\right] \tag{11}$$

which may be termed the similarity-law roughness characterization. For smooth walls $\Delta B = 0$. The functional dependence of ΔB arises naturally from the overlapping of the inner and the outer laws. The ΔB characterization may be obtained empirically for any type of roughness $\tilde{R}$ and is a key element in the prediction of the drag augmentation of hydrodynamic bodies due to rough surfaces. In general for rough surfaces ΔB is negative.

An alternate expression of the inner law is

$$\frac{u}{u_\tau} = A \ln \frac{y}{k} + B_r\left[k^*, \tilde{R}\right]$$

where

$$B_r = B_1 + A \ln k^*$$

For the case of fully-developed roughness flow B_r is constant with value $B_{r,0}$ and

$$\Delta B = B_{r,0} - B_{1,0} - A \ln k^*$$

ΔB is then a straight line for fully-developed roughness flow on a plot of ΔB against $\ln k^*$.

For the outer law, the logarithmic velocity law becomes

$$\frac{U-u}{u_\tau} = -A \ln \frac{y}{\delta} + B_2$$

which is the same as that for smooth surfaces.

EXPERIMENTAL DETERMINATIONS OF ROUGHNESS CHARACTERIZATIONS

General

For a particular roughness geometry $\tilde{R}$, the similarity-law roughness characterization ΔB has to be obtained empirically as a function of a roughness Reynolds number k^*. A direct determination requires a velocity survey of the inner region of the shear layer which is expressed as

$$\left(\frac{u}{u_\tau}\right)_r = A \ln y^* + B_{1,0} + \Delta B \tag{16}$$

A smooth surface gives

$$\left(\frac{u}{u_\tau}\right)_s = A \ln y^* + B_{1,0} \tag{17}$$

At the same y^* then

$$\Delta B = \left(\frac{u}{u_\tau}\right)_r - \left(\frac{u}{u_\tau}\right)_s \tag{18}$$

The direct velocity method was used by Nikuradse[3] in his classical flow measurements for dense sand-grain roughness in a pipe flow. Since such velocity measurements may be difficult and arduous to perform, it is more convenient to resort to indirect methods requiring just overall measurements, such as the average velocity and pressure drop for pipe flow, the total drag of a flat plate, and the torque of a rotating disk.

Pipe Flow

In fully-developed pipe flow, it is more convenient to determine the average velocity from the rate of mass flow than to perform a velocity survey. The similarity laws may then be used to relate the ΔB-characterization to the average velocity.

In general, for straight circular pipes with fully-developed turbulent flow (no entrance effects), the friction loss or pressure drop represented by wall shear stress τ_w may be related to an average velocity V of the pipe flow, pipe diameter D, fluid properties of density ρ and kinematic viscosity ν, and a geometrical description of the irregular rough wall $k, k_1, k_2, \ldots,$ which symbolically becomes

$$\tau_w = f\left[V, D, \rho, \nu, k, k_1, k_2, \cdots\right] \tag{19}$$

A dimensional analysis results in a friction factor f as a function of two dimensionless ratios, a Reynolds number R_D and a relative roughness D/k for a particular roughness configuration $\underset{\sim}{R}$ or

$$f = f\left[R_D, \frac{D}{k}, \tilde{R}\right] \tag{20}$$

where

the Fanning friction factor f is given by

$$f = \frac{\tau_w}{\frac{1}{2}\rho V^2} \tag{21}$$

and the pipe Reynolds number by

$$R_D = \frac{VD}{\nu} \tag{22}$$

The Darcy-Weisbach friction factor λ may also be used where

$$\lambda = 4f \tag{23}$$

The ΔB-characterization may be related to overall factors as follows. For a circular pipe, the average velocity V is given by

$$V = \frac{2\pi \int_0^\delta u(\delta - y) y \, dy}{\pi \delta^2} \tag{24}$$

where $\delta = D/2$, pipe radius

which in inner law variables becomes

$$\frac{V}{u_\tau} = \frac{2}{\eta}\left(\int_0^\eta \frac{u}{u_\tau} dy^* - \frac{1}{\eta}\int_0^\eta \frac{u}{u_\tau} y^* dy^*\right) \tag{25}$$

where $\eta = \frac{u_\tau \delta}{\nu}$.

Application of both velocity similarity laws and neglect of the very thin laminar and buffer sublayers results in

$$\frac{V}{u_\tau} = A \ln \eta + B_{1,0} + \Delta B - \frac{43}{30} A + \frac{3}{10} B_2 \tag{26}$$

Since by definition

$$\frac{V}{u_\tau} = \sqrt{\frac{2}{f}} \tag{27}$$

and

$$\eta = \frac{\sqrt{2}}{4}\sqrt{f}\, R_D \tag{28}$$

then for rough surfaces

$$\left(\frac{1}{\sqrt{f}}\right)_r = \frac{A}{\sqrt{2}} \ln \sqrt{f}\, R_D + \frac{1}{\sqrt{2}}\left(B_{1,0} + \Delta B - \frac{43}{30}A + \frac{3}{10}B_2 - \frac{A}{2}\ln 8\right) \tag{29}$$

For smooth surfaces $\Delta B=0$ and then

$$\left(\frac{1}{\sqrt{f}}\right)_s = \frac{A}{\sqrt{2}} \ln \sqrt{f}\, R_D + \frac{1}{\sqrt{2}}\left(B_{1,0} - \frac{43}{30}A + \frac{3}{10}B_2 - \frac{A}{2}\ln 8\right) \tag{30}$$

For the same value of $\sqrt{f}\, R_D$

$$\Delta B = \left(\sqrt{\frac{2}{f}}\right)_r - \left(\sqrt{\frac{2}{f}}\right)_s \tag{31}$$

and the corresponding k^* is given by

$$k^* = \left(\sqrt{f}\, R_D\right)\frac{k}{\sqrt{2}\, D} \tag{32}$$

which follows from definitions.

The procedure for determining $\Delta B[k^*,\tilde{R}]$ from pipe flow is then as follows.

1. For roughness configuration $\tilde{R}$ inside a pipe of diameter D plot the measured data reduced to dimensionless ratios f and R_D in coordinates $\frac{1}{\sqrt{f}}$ versus log $\sqrt{f}\, R_D$. Then plot the smooth friction line in the same coordinate system.

2. At specific values of abscissa $\sqrt{f}\,R_D$ determine ΔB from Equation (31) and k^* from Equation (32).

Flat Plate

For the case of a towed flat plate with an irregular rough surface, it is certainly more convenient to measure the drag and towing velocity than to measure the local velocity profile in a thin boundary layer and the accompanying local wall shear stress. Indirectly the similarity laws may then be used to determine the ΔB-characterization as a function of roughness Reynolds number k^* from the total drag and towing velocity.

In general, the drag $\tilde{D}$ of a rough flat plate towed parallel to its motion depends on forward velocity U_∞, surface area S, length L, fluid properties of density ρ and kinematic viscosity ν and a geometrical description of its irregular rough wall k, k_1, k_2,... which symbolically becomes

$$\tilde{D} = f\left[U_\infty, S, L, \rho, \nu, k, k_1, k_2, \cdots\right] \tag{33}$$

A dimensional analysis results in a drag coefficient C_F as a function of two dimensionless ratios, a Reynolds number R_L and a relative roughness L/k for a particular roughness configuation $\tilde{R}$ or

$$C_F = f\left[R_L, \frac{L}{k}, \tilde{R}\right] \tag{34}$$

where drag coefficient

$$C_F = \frac{\tilde{D}}{\frac{1}{2}\rho U_\infty^2 S} \tag{35}$$

and Reynolds number

$$R_L = \frac{U_0 L}{\nu} \tag{36}$$

The ΔB-characterization, may be obtained indirectly from drag measurements of a rough surface by considering deviations from a smooth friction line as follows.

An application of the similarity laws results in a logarithmic formula for the drag of rough flat plates

$$\ln R_L C_F = \frac{\sqrt{2}}{A}\frac{1}{\sqrt{C_F}} + 1 - \frac{B_2}{A} - \frac{B_{1,0}}{A} - \frac{(\Delta B)_e}{A} + \ln 2 D_1 \tag{37}$$

Here D_1 is a boundary-layer constant defined in Reference 6 and $(\Delta B)_e$ is the value of ΔB for the boundary layer at the end of the flat plate. For a smooth wall $\Delta B = 0$. Then

$$\ln R_L C_F = \frac{\sqrt{2}}{A}\frac{1}{\sqrt{C_F}} + 1 - \frac{B_2}{A} - \frac{B_{1,0}}{A} + \ln 2 D_1 \tag{38}$$

Consider a plot of $\frac{1}{\sqrt{C_F}}$ against log $R_L C_F$ for both rough and smooth plates. Then at the same values of $R_L C_F$

$$(\Delta B)_e = \left(\sqrt{\frac{2}{C_F}}\right)_r - \left(\sqrt{\frac{2}{C_F}}\right)_s \tag{39}$$

The corresponding k_e^* at the end of the flat plate is obtained as follows: From definitions

$$k_e^* = \left(\frac{u_\tau}{U_\infty}\right)_e R_L \left(\frac{k}{L}\right) \tag{40}$$

and the corresponding local friction

$$\left(\frac{u_\tau}{U_\infty}\right)_e = \sqrt{\frac{C_F}{2}}\left(1 - A\sqrt{\frac{C_F}{2}}\right) \tag{41}$$

The procedure for determining $\Delta B[k^*, \tilde{R}]$ is then as follows:

1. For roughness configuration $\tilde{R}$ on a flat plate of length L plot the measured data (C_F, R_L) in coordinates $\frac{1}{\sqrt{C_F}}$ against log $R_L C_F$. Plot the smooth friction line in the same coordinate system.

2. At specific values of abscissa $R_L C_F$, determine ΔB from Equation (39) and k^* from Equations (40) and (41).

Rotating Disk

Although the boundary-layer flow on a rotating disk is three dimensional in contrast to the two-dimensional flow on a flat plate or through a pipe, the rotating disk provides a convenient method of characterizing roughness effects particularly at high speeds. Determining the ΔB-characterization directly by a velocity survey of the boundary layer of a rotating disk is physically feasible but definitely more arduous than determining ΔB indirectly from overall measurements.

In general the torque or moment M of one side of a rotating disk in an unbounded fluid medium depends on the angular velocity ω, the radius R, fluid properties of density ρ and kinematic viscosity ν, and the geometry of the irregular rough wall k, k_1, k_2... or symbolically

$$M = f\left[\omega, R, \rho, \nu, k, k_1, k_2, \cdots\right] \tag{42}$$

A dimensional analysis results in a moment coefficient C_m as a function of two dimensionless ratios, a Reynolds number R_R and a relative roughness R/k for a particular roughness configuration $\tilde{R}$ or

$$C_m = f\left[R_R, \frac{R}{k}, \tilde{R}\right] \tag{43}$$

where moment coefficient

$$C_m = \frac{4M}{\rho R^5 \omega^2} \tag{44}$$

and Reynolds number

$$R_R = \frac{\omega R^2}{\nu} \tag{45}$$

The ΔB-characterization may be obtained indirectly from overall measurements as follows. The application of the similarity laws results in a logarithmic formula[8] for rough surfaces

$$\frac{1}{\sqrt{C_m}} = A\sqrt{\frac{5}{8\pi}} \ln R_R\sqrt{C_m} + \sqrt{\frac{5}{8\pi}}\left[B_{1,0} + (\Delta B)_e\right] - \frac{A}{\sqrt{10\pi}} - \\ - A\sqrt{\frac{5}{8\pi}} \ln \frac{55}{18}\sqrt{4\pi}\,A^2 - \frac{(\Delta B)'_e}{\sqrt{40\pi}} \tag{46}$$

where $(\Delta B)_e$ and $(\Delta B)'_e$ are the values of ΔB and $(\Delta B)'$ at the edge of the disk, and

$$(\Delta B)' = \frac{d\Delta B}{d\ln k^*} \tag{47}$$

For a smooth surface $\Delta B = 0$ and $(\Delta B)' = 0$.

Then

$$\frac{1}{\sqrt{C_m}} = A\sqrt{\frac{5}{8\pi}} \ln R_R\sqrt{C_m} + \sqrt{\frac{5}{8\pi}}\,B_{1,0} - \frac{A}{\sqrt{10\pi}} - \\ - A\sqrt{\frac{5}{8\pi}} \ln \frac{55}{18}\sqrt{4\pi}\,A^2 \tag{48}$$

Consider a plot of $\frac{1}{\sqrt{C_m}}$ against log $R_R\sqrt{C_m}$ for both rough and smooth surfaces. Then for the same value of $R_R\sqrt{C_m}$

$$(\Delta B)_e = \sqrt{\frac{8\pi}{5}}\left[\left(\frac{1}{\sqrt{C_m}}\right)_r - \left(\frac{1}{\sqrt{C_m}}\right)_s\right] + \frac{(\Delta B)'_e}{5} \tag{49}$$

The corresponding value of k^*_e at the edge of the disk is obtained as follows

From definitions

$$k^*_e = \left(\frac{u_\tau}{\omega R}\right)_e R_R \left(\frac{k}{R}\right) \tag{50}$$

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[8] Granville, P.S., "The Torque and Turbulent Boundary Layer of Rotating Disks with Smooth and Rough Surfaces, and in Drag-Reducing Polymer Solutions," Journal of Ship Research, Vol. 17, No. 4, pp. 181-195 (Dec 1973).

and the corresponding local friction at the edge of the disk is

$$\left(\frac{u_\tau}{\omega R}\right)_e = \sqrt{\frac{5}{8\pi}}\sqrt{C_m}\left\{1 - \left[2A + (\Delta B)'_e\right]\sqrt{\frac{1}{40\pi}}\sqrt{C_m}\right\} \tag{51}$$

The procedure for determining $\Delta B\ [k^*, \tilde{R}]$ is then as follows.

1. For roughness configuration $\tilde{R}$ on a disk of radius R plot the measured data (C_m, R_R) in coordinates $\frac{1}{\sqrt{C_m}}$ against log $R_R \sqrt{C_m}$. Plot the smooth friction line in the same coordinates.

2. At specific values of abscissa $R_R \sqrt{C_m}$, determine ΔB from Equation (49) and k* from Equations (50) and (51). The effect of $(\Delta B)^1$ may be obtained by reiteration. Application to enclosed rotating disks is shown in the Appendix.

A check on the validity of the similarity laws for rotating disks is given by the excellent agreement of the torque of a rotating disk covered with dense sand grains with similarity-law predictions for the fully rough regime[9].

APPLICATION OF SIMILARITY-LAW ROUGHNESS CHARACTERIZATIONS TO PREDICTING VISCOUS DRAG

Boundary-Layer Calculations

Since the viscous drag of bodies may be determined from the boundary-layer development,[10] the increased drag due to roughness may also be determined by applying the roughness characterization to a turbulent boundary-layer prediction method. In general there are now two principal turbulent boundary-layer methods:

1. "Differential" methods for solving the boundary-layer equations of motion directly by using a turbulent shear stress model such as that of eddy viscosity or mixing length.

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[9] Dorfman, L.A., "Hydrodynamic Resistance and the Heat Loss of Rotating Solids," Oliver and Boyd, Edinburgh, 1963.

[10] Granville, P.S., "The Calculation of the Viscous Drag of Bodies of Revolution," David Taylor Model Basin Report 849 (Jul 1963).

2. "Integral" methods for solving the integrated boundary-layer equations of motion by using relations for velocity profile and local skin friction. To date, no "differential" method has been published which includes roughness effects although Cebeci and Smith[11] allude to a possible method. The effect of roughness has been included in an "Integral" method which extends the two-dimensional empirical Head entrainment method by altering the local skin friction.[12] The effect of roughness may also be readily included in a similarity-law entrainment method for two-dimensional and axisymmetric boundary layers.[7,13]

Form Factors for Rough Surfaces

For streamlined bodies for which no appreciable flow separation occurs a simplified method has been developed[14] for separating the effect of shape and of roughness on viscous drag. A form factor is defined which accommodates the effect of shape while an equivalent flat plate is defined which takes care of roughness effects.

In general the drag of a body D_v depends on the forward speed U_∞, the surface area S, the density ρ and kinematic viscosity ν of the fluid, the forward length L, and other lengths defining the shape L_1, L_2,... and the geometry of the irregular roughness k, k_1, k_2... or

$$D_v = f\left[U_\infty, S, \rho, \nu, L, L_1, L_2, \cdots, k, k_1, k_2, \cdots\right] \qquad (52)$$

A dimensional analysis results in

$$C_v = f\left[R_L, \tilde{S}, \frac{k}{L}, \tilde{R}\right] \qquad (53)$$

where drag coefficient C_v is given by

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[11] Cebeci, T. and A.M.O. Smith, "Analysis of Turbulent Boundary Layers," Academic Press, New York, 1974.

[12] Dvorak, F.A., "Calculation of Turbulent Boundary Layers on Rough Surfaces in Pressure Gradient," AIAA Journal, Vol. 7, No. 9, pp. 1752-1759 (Sep 1969).

[13] Granville, P.S., "Similarity-Law Entrainment Method for Thick Axisymmetric Turbulent Boundary Layers in Pressure Gradients," David Taylor Naval Ship R&D Center Report 4525 (Dec 1975).

$$C_v = \frac{D_v}{\frac{1}{2}\rho U_\infty^2 S} \tag{54}$$

the Reynolds number by

$$R_L = \frac{U_\infty L}{\nu} \tag{55}$$

the shape of the body by length ratios

$$\tilde{S} = \frac{L}{L_1}, \frac{L_1}{L_2}, \ldots \tag{56}$$

and the shape of the irregular roughness by length ratios

$$\tilde{R} = \frac{k}{k_1}, \frac{k_1}{k_2}, \ldots \tag{57}$$

A form factor $\tilde{f}$ is defined so that

$$C_v\left[R_L, \tilde{S}, \frac{k}{L}, \tilde{R}\right] = \left(1 + \tilde{f}\left[\tilde{S}\right]\right) C_F\left[R_L, \frac{k}{L}, \tilde{R}\right] \tag{58}$$

where C_F is the equivalent flat-plate drag coefficient. An equivalent flat plate is defined as a flat plate with the same surface area and length as the body.

By a simplified boundary-layer analysis it may be shown[14] that a form factor $\tilde{f}$ may be determined approximately from the pressure along the body represented by velocity ratio $\frac{U}{U_\infty}$. Here U is the velocity outside the boundary layer.

For two-dimensional foils at zero angle of attack[14]

$$1 + \tilde{f} = \left[\frac{\int_0^1 \left(\frac{U}{U_\infty}\right)^{H+2} \sec\alpha \, d\left(\frac{x}{L}\right)}{\int_0^1 \sec\alpha \, d\left(\frac{x}{L}\right)}\right] \left(\frac{U}{U_\infty}\right)_e^{\frac{1-H}{1+q}} \tag{59}$$

[14] Granville, P.S., "A Prediction Method for the Viscous Drag of Ships and Underwater Bodies with Surface Roughness and/or Drag-Reducing Polymer Solutions," David Taylor Naval Ship R&D Center Report SPD-797-01 (Oct 1977).

Here H is a two-dimensional shape parameter

x is a chord distance from the nose

q is a wake factor

e is a subscript representing conditions at the tail end and

α is an angle of the tangent of the contour relative to the chord

For bodies of revolution in axisymmetric flow[14]

$$1+\tilde{f} = \left[\frac{\int_0^1 \left(\frac{r}{L}\right)\left(\frac{U}{U_\infty}\right)^{h+2} \sec\alpha \, d\left(\frac{x}{L}\right)}{\int_0^1 \left(\frac{r}{L}\right) \sec\alpha \, d\left(\frac{x}{L}\right)}\right] \left(\frac{U}{U_\infty}\right)_e^{\frac{1-h}{1+q}} \tag{60}$$

Here h is an axisymmetric shape parameter

r is a radius of cross-section of the body of revolution

x is an axial distance from the nose

and α is an angle of the tangent of the meridian contour relative to the axis

For ship hulls[12] two equivalent bodies of revolution are used in Equation (60) to give

$$1+\tilde{f} = \left[\frac{\int_0^1 \left(\frac{r_p}{L}\right)\left(\frac{U}{U_\infty}\right)_A^{h+2} \sec\alpha \, d\left(\frac{x}{L}\right)}{\int_0^1 \left(\frac{r_p}{L}\right) \sec\alpha \, d\left(\frac{x}{L}\right)}\right] \left(\frac{U}{U_\infty}\right)_{A,e}^{\frac{1-h}{1+q}} \tag{61}$$

Here r_p is a radius of a cross-section of an equivalent body of revolution based on perimeter. $\left(\frac{U}{U_\infty}\right)_A$ is a velocity ratio of an equivalent body of revolution based on cross-sectional area defined by r_A. Also

$$r_p = \frac{\tilde{P}}{\pi} \tag{62}$$

and

$$r_A = \sqrt{\frac{2\tilde{A}}{\pi}} \tag{63}$$

where

P is the local perimeter of a ship's underwater hull.

$\tilde{A}$ is a cross-sectional area of a ship underwater hull at an axial station.
A constant value of H such as H = n = 1.4 is recommended although other values of H could accommodate any changes in form factor $\tilde{f}$ due to viscous effects. For two-dimensional foils q = 1 and for bodies of revolution q = 7.

FLAT-PLATE DRAG DIAGRAMS

Flat-Plate Drag Diagrams from Roughness Characterizations

For a given irregular roughness configuration $\tilde{R}$ the roughness characterization ΔB is a function of a single dimensionless ratio, Reynolds number k^*. The requirement is to convert ΔB to a flat-plate drag coefficient C_F as a function of two dimensionless ratios, Reynolds number R_L and relative roughness L/k. A convenient method to determine each point $C_F[\log_{10} R_L, L/k]$ is what may be termed the method of the interaction of two loci: one locus to satisfy ΔB and another to satisfy k^*. These may be performed graphically. The first locus to satisfy ΔB depends on a smooth friction line[15] such as

$$(C_F)_s = \frac{0.0776}{(\log_{10} R_L - 1.88)^2} + \frac{60}{R_L} \tag{64}$$

which agrees with the well-known Schoenherr line at high Reynolds numbers but is more accurate at low Reynolds number. From Equation (37) for the same values of C_F and in common logarithms

[15] Granville, P.S., "The Drag and Turbulent Boundary Layer of Flat Plates at Low Reynolds Numbers," Journal of Ship Research, Vol. 21, No. 1, pp. 30-39 (Mar 1977).

$$\log_{10}(R_L)_r - \log_{10}(R_L)_s = -\frac{(\Delta B)_e}{2.3A} \tag{65}$$

Hence, for constant ΔB the locus for the friction line with roughness is offset by a constant amount in the $\log_{10} R_L$ direction.

The second locus which satisfies k^* is obtained by taking the common logarithm of Equation (40) and by utilizing Equation (42) so that

$$\log_{10} R_L = \log_{10} k^* + \log_{10}\sqrt{\frac{2}{C_F}} + \frac{A}{2.3}\sqrt{\frac{C_F}{2}} + \log_{10}\frac{L}{k} \tag{66}$$

Each value of relative roughness L/k produces another locus.

The intersection of the two loci produces a point $(C_F, \log_{10} R_L)$ for a particular L/k and $\tilde{R}$. Other points are determined to define a friction line C_F $[\log_{10} R_L, L/k]$ for roughness configuration $\tilde{R}$.

Roughness Friction Lines for Different Flat-Plate Lengths

If a friction line for a particular roughness $\tilde{R}$ is known for one length, it is often necessary to obtain friction lines for other flat-plate lengths. To find C_F one way would be to determine the ΔB-correlation and use the method of two loci previously described. A more direct procedure is now described which also involves two loci.

Each point on a given roughness friction line for a plate of length L_1 represents a value of ΔB and k^*. The object is to obtain a point on a new roughness friction line for a different length plate L_2 with the same values of ΔB and k^*. The locus of a line with the same ΔB is given by Equation (37)

$$\log_{10}(R_L)_r - \log_{10}(R_L)_s = -\frac{\Delta B}{2.3A}$$

which shows that the locus is offset a fixed distance from the smooth friction line in the $\log_{10} R_L$ direction.

The other locus for the same value of k^* but with a different length L_2 is obtained as follows. Consider a combined factor L^* where by definitions

$$L^* = k^*\left(\frac{L}{k}\right) = \frac{u_\tau L}{\nu} = \left(\frac{u_\tau}{U_\infty}\right) R_L$$

With $\dfrac{u_\tau}{U_\infty}$ given by Equation (42)

$$R_L = \frac{L^*}{\sqrt{\frac{C_F}{2}}\left(1 - A\sqrt{\frac{C_F}{2}}\right)}$$

For the same values of C_F and same k^*

$$\log_{10} R_{L,2} - \log_{10} R_{L,1} = \log_{10} \frac{L_2^*}{L_1^*} = \log_{10} \frac{L_2}{L_1}$$

Here $R_{L,1}$ and $R_{L,2}$ are defined for L_1 and L_2 respectively. The procedure is to draw a reference locus for a convenient value of L^* to satisfy Equation (68). Through point P_1 on the friction line for L_1 a locus is drawn which is offset a constant distance in the log R_L direction. The locus for the same k^* and new length L_2 is offset a constant distance given by $\log_{10} \frac{L_2}{L_1}$. The intersection of the locus for constant L^* and that for constant k^* gives point P_2 for the new friction line corresponding to length L_2 as shown in Figure 6.

NUMERICAL EXAMPLES

To illustrate the procedures described here some typical cases are examined. The first case to be considered is that of a 9-inch diameter rotating disk with a random roughness machined on both sides with a height of about 0.005 inch.[16] The C_m, R_R data are plotted in the form of $\frac{1}{\sqrt{C_m}}$ against log $R_R\sqrt{C_m}$ in Figure 1 for both rough and smooth disks. At the same values of $R_R\sqrt{C_m}$, the roughness characterization ΔB is obtained from Equation (40) with $(\Delta B)^1$ considered zero at first. The corresponding k^* is obtained from Equations (50) and (51) and the results are plotted in

[16] Amfilokhiev, W.B. and A.M. Ferguson, "The Variation of Friction Drag with Surface Roughness in Dilute Polymer Solutions," University of Glasgow, Department of Naval Architecture Experiment Tank Report [illegible] (Aug [illegible]).

Figure 2. The slope of the correlating line gives the values of $(\Delta B)'$ which is then used to recalculate the values of ΔB as given by Equation (49). The reiterated final results are also plotted in Figure 2.

The preparation of a flat-plate roughness diagram, C_F, R_L, is shown in Figure 3 for L = 300 feet and L = 600 feet. The curve for the smooth surface is plotted from values calculated from Equation (64). Selected points (ΔB, k*) of the roughness correlation of Figure 2 are treated as follows. In accordance with Equation (65) a locus of constant ΔB is drawn offset a value of $-\frac{\Delta B}{2.3A}$ from the smooth curve (A = 2.5). A locus of constant k* for L = 300 feet and L = 600 feet are plotted to satisfy Equation (66) by calculating R_L as a function of C_F. The intersections give points on the lines for L = 300 feet and L = 600 feet respectively. The procedure is repeated for other values of ΔB and k*. The resulting curves are plotted in Figure 3 within the limits of the experimental data. The largest value of Reynolds number R_L for L = 300 feet is $R_L = 1.8 \times 10^9$ which corresponds to a speed of about 43 knots which is higher than the operating velocity of most vessels. This shows that the rotating disk easily provides results applicable to full-scale speeds.

The next case to be considered is roughness on a towed flat plate, a 21-foot friction plane [2]. The roughness was cast from the bottom plating of a ship hull which had been grit blasted to bare metal and painted with vinyl paint. It was designated as Plate C in Reference 2. It was very rough with a roughness height of about 0.03 inch. The drag data is plotted in Figure 4 as $\sqrt{\frac{2}{C_F}}$ against log $R_L C_F$. the ΔB-characterization is calculated from Equation (40) from values of the difference in $\sqrt{\frac{2}{C_F}}$ between the rough and smooth plates at the same value of $R_L C_F$. The corresponding values of k* are determined by using Equations (41) and (42). The results are shown in Figure 5 for the ΔB-k* correlation.

Although the flat-plate roughness diagram can be calculated from the ΔB-characterization, a more direct extrapolation method for the towed flat-plate drag data will be used to illustrate the method of preparing friction lines for different lengths of flat plate. The locus of a constant ΔB is shown in Figure 6 as a constant offset of the smooth friction line in

accordance with Equation (37). A reference line for constant L* is then shown satisfying Equation (68) by selecting a convenient value of L*. The loci of lines of constant k* are shown offset from the reference line by a constant distance in accordance with Equation (69). The procedure is repeated for selected values of the measured friction line. Two lengths of 300 feet and 600 feet are shown as an illustration. The maximum value of R_L for the 300-foot plate is 7.2×10^8 which corresponds to a speed of about 17 knots. For higher full-scale speeds, it is then necessary to increase the speed of the towed friction plane to provide extrapolations for operating ship speeds.

CONCLUDING REMARKS

The three methods described here of characterizing the drag increase due to roughness have various advantages and disadvantages. Hydrodynamically, fully-developed turbulent pipe flow provides constant local conditions in contrast to the variable turbulent boundary layers on flat plates and rotating disks. The three-dimensional boundary layers on rotating disks are more complicated than the two-dimensional boundary layers on flat plates.

The difficulty of applying irregular roughnesses to the inside of pipes may require pipe test sections split longitudinally for easy access. In addition, pipes have to be long enough to attain fully developed pipe flow. Since the wall shear stress results in a very small longitudinal pressure gradient, the pipes have to be very straight and very level for accurate measurements. If the roughness applied to the inside of a pipe has appreciable thickness, the new effective inside diameter may have to be accurately determined.

A major problem with towed flat plates in basins is obtaining a sufficiently high enough speed without undue free-surface effects. However this is not a problem with plates in a water tunnel. On the other hand high peripheral speeds are easily attained on a rotating disk. In addition, a rotating-disk apparatus is usually a one-purpose equipment in constrast to multi-purpose towing carriages or water tunnels used for flat plates which compete for time with other hydrodynamic testing. It would be reassuring to conduct definitive experiments to verify that the same ΔB-characterizations result irrespective of method.

APPENDIX - Enclosed Rotating Disk

For an enclosed rotating disk[17] the effective angular velocity is reduced to $\phi\omega$ where ϕ is a factor accounting for the swirl of the enclosed fluid, $0<\phi\leq 1$. C_m and R_R are modified accordingly in Equations (44) and (45) for use in Equations (46) through (51) as follows

$$C_m = \frac{4M}{\rho R^5 (\phi\omega)^2} \tag{70}$$

and

$$R_R = \frac{\phi\omega R^2}{\nu} \tag{71}$$

The factor ϕ may be experimentally determined by a comparison of the smooth lines of $\frac{1}{\sqrt{C_m}}$ plotted against $\log R_R\sqrt{C_m}$ for enclosed and unenclosed rotating disks. Since ϕ does not affect the product $R_R\sqrt{C_m}$, then from definitions

$$\phi = \frac{\left(\frac{1}{\sqrt{C_m}}\right)_{en}}{\left(\frac{1}{\sqrt{C_m}}\right)_{\infty}} \tag{72}$$

where subscripts en and ∞ refer to enclosed and unenclosed conditions respectively. The factor ϕ may be found[17] to be a function of $R_R\sqrt{C_m}$.

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[17] Daily, J.W. and R.E. Nece, "Chamber Dimension Effects on Induced Flow and Frictional Resistance of Enclosed Rotating Disks," Journal of Basic Engineering (Transactions of ASME), Vol. 82, Series D, pp. 217-232 (Mar 1960).

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Figure 1 - Moment Data for Rotating Disk

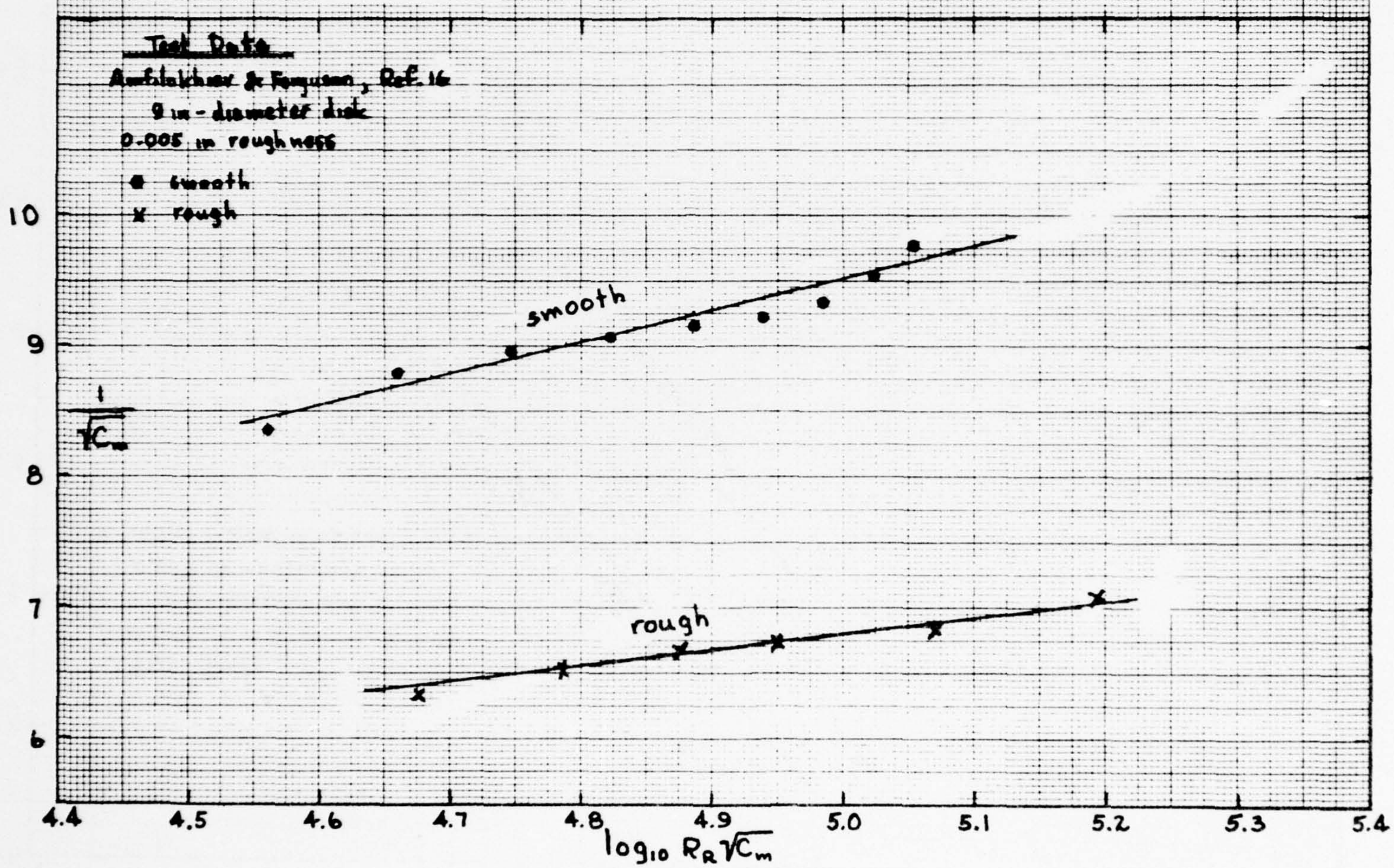
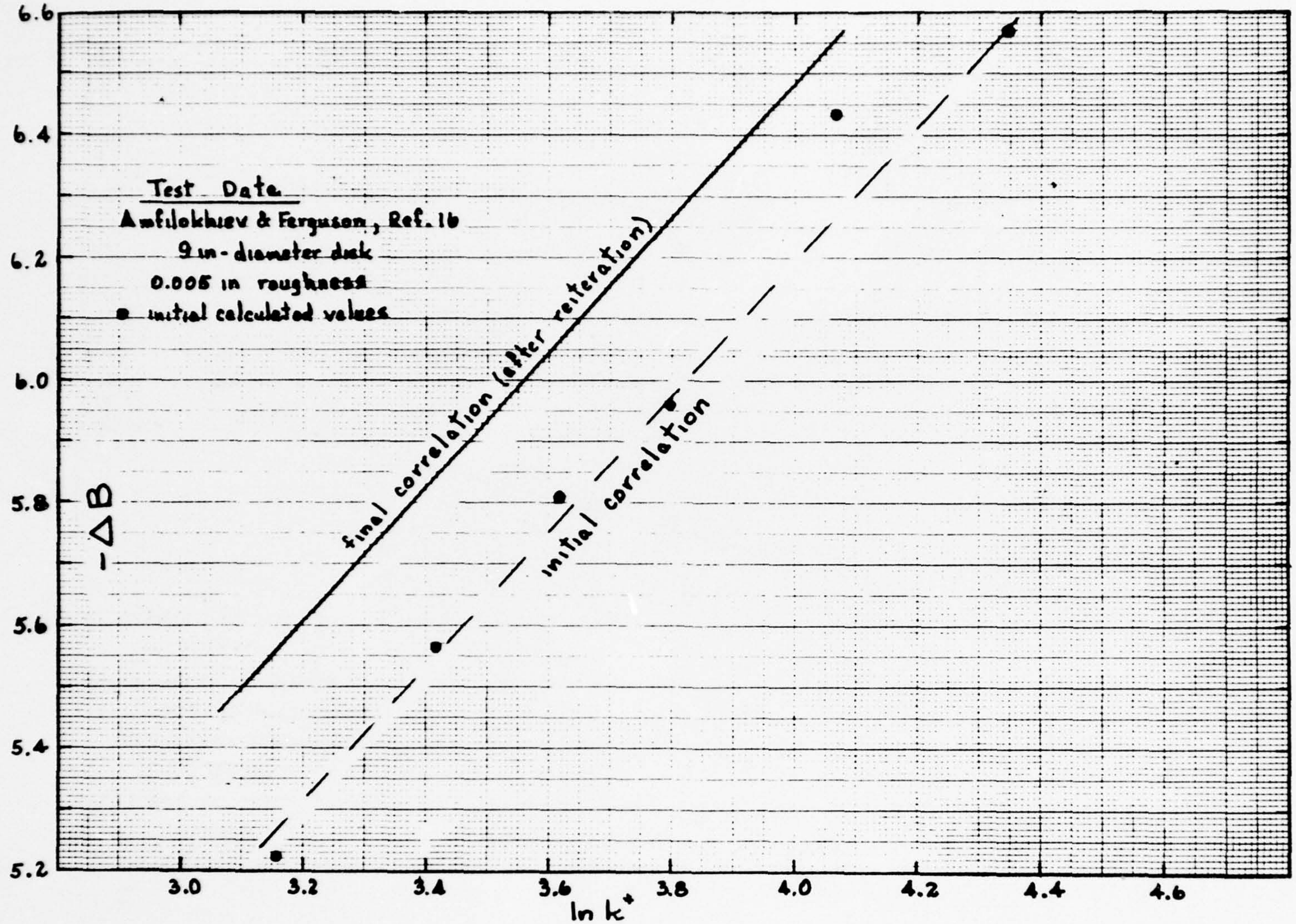


Figure 2 - ΔB-Correlation from Rotating Disk



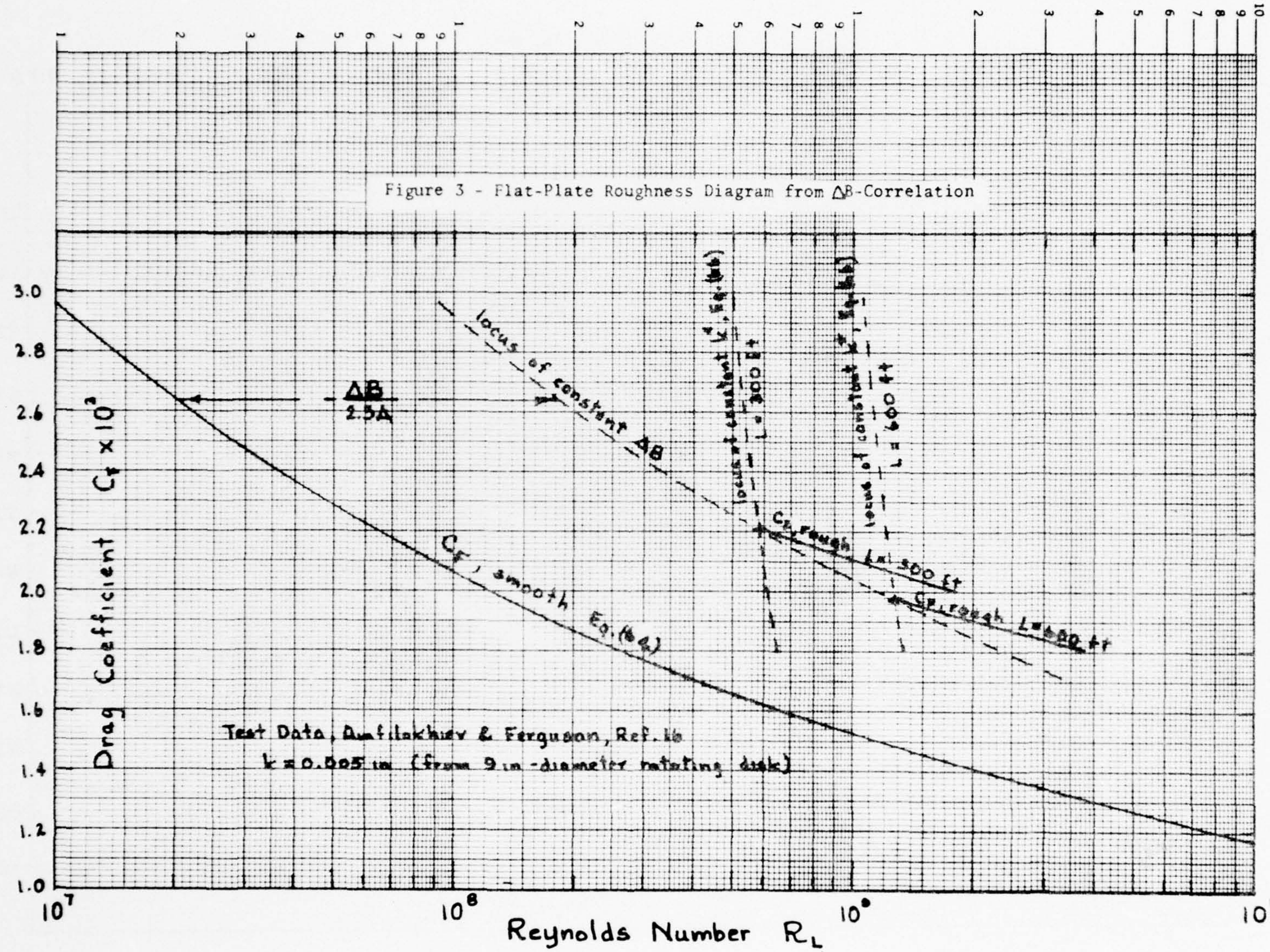


Figure 3 - Flat-Plate Roughness Diagram from ΔB-Correlation

Figure 4 - Drag Data for Towed Flat Plate

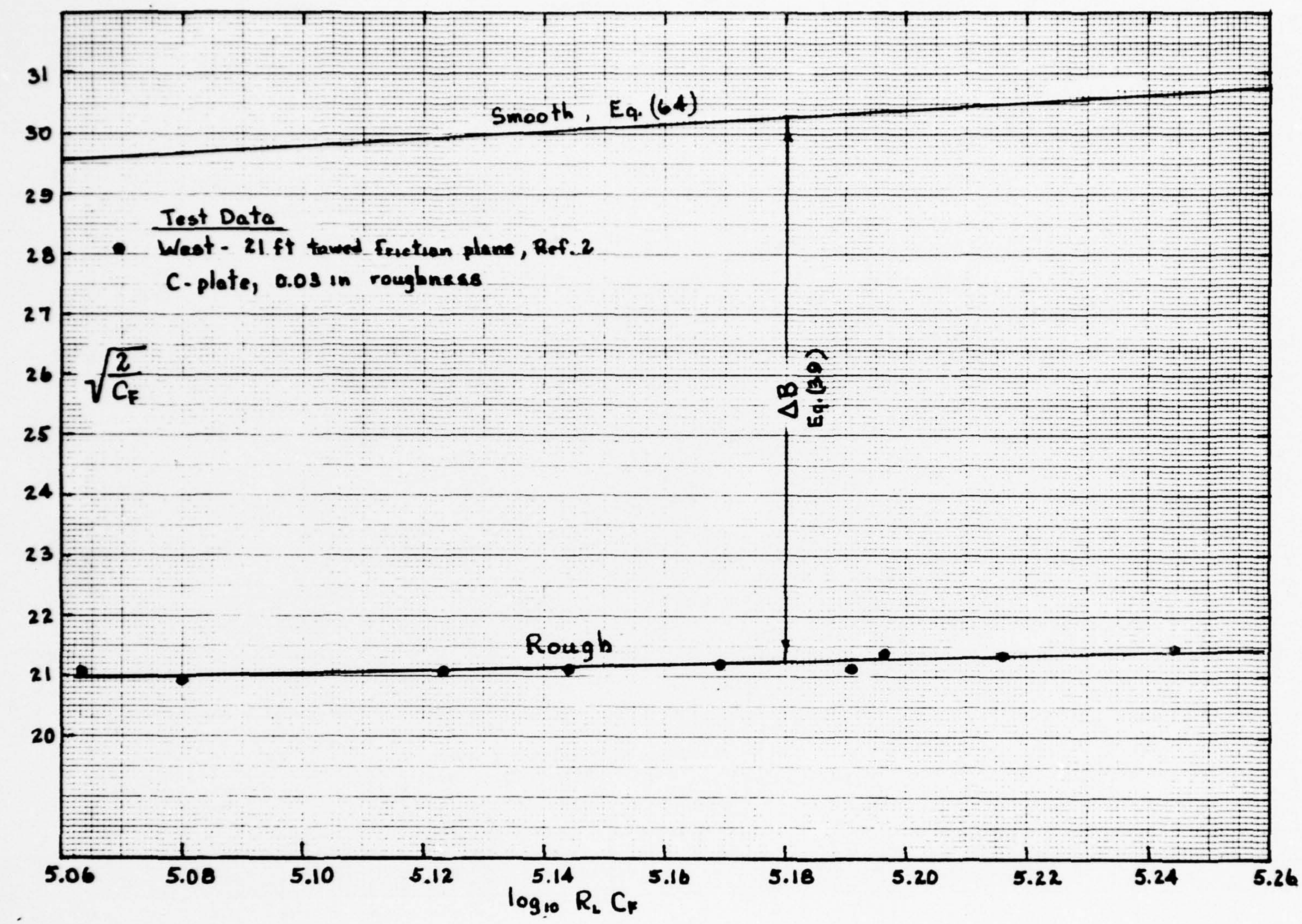
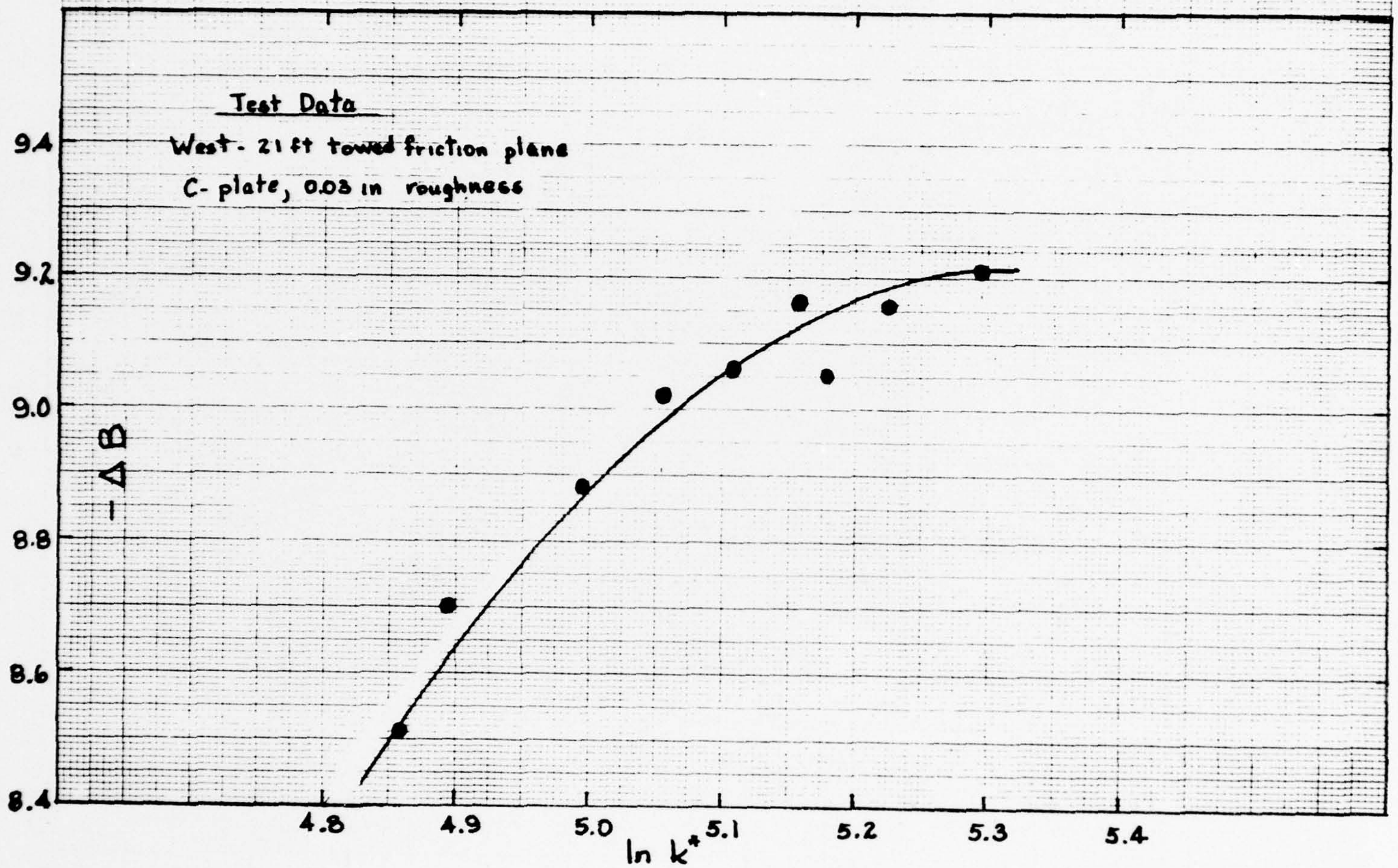


Figure 5 - ΔB-Correlation from Towed Flat Plate



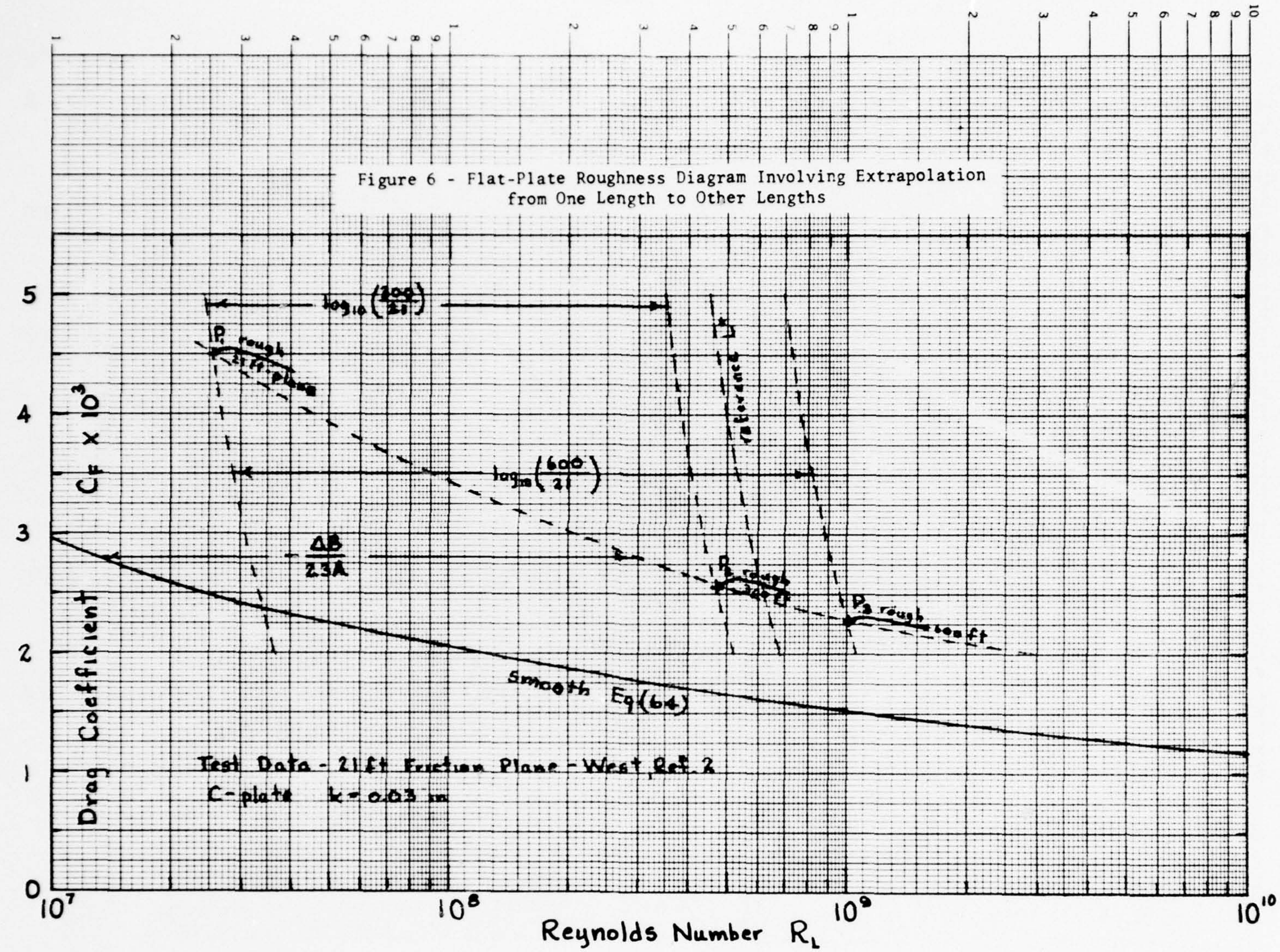


Figure 6 - Flat-Plate Roughness Diagram Involving Extrapolation from One Length to Other Lengths

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